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IN THE UNITED STATES PATENT AND TRADEMARK OFFICE

Applicant	:	Yoshioka, et al.
Appl. No.	:	10/687,710
Filed	:	October 20, 2003
Title	:	SOLENOID ACTUATED VARIABLE PRESSURE RELIEF VALVE ASSEMBLY FOR TORQUE TRANSFER ASSEMBLY
Group Art Unit	:	3681
Examiner	:	Tisha LEWIS
Docket No.	:	08200.836

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APPEAL BRIEF UNDER 37 C.F.R. § 41.37

January 17, 2006

Hon. Commissioner of Patents
and Trademarks
PO Box 1450
Alexandria, VA 22313-1450

Dear Sir:

In follow-up to the Notice of Appeal filed November 14, 2005, Appellant respectfully requests the Board of Patent Appeals and Interferences consider the following arguments and reverse the decision of the Examiner in whole.

Applicant has submitted herewith the fee prescribed by 37 CFR 41.20(b)(2).

No additional fees are deemed necessary at this time; however, the Commissioner is hereby authorized to charge applicant's deposition account no. 50-0548 to maintain the pendency of this application.

(1) Real Party in Interest

The real party in interest is TORQUE-TRACTION TECHNOLOGIES, INC. - a division of Dana Corporation.

(2) Related Appeals and Interferences

There are no known related appeals or interferences, which will directly affect or be directly affected by or have a bearing on the decision in the pending appeal.

(3) STATUS OF CLAIMS

1. Claims 1-25 are pending in the application.
2. Claims 1 and 11-23 have been rejected and are being appealed.
3. Claims 2-10, 24 and 25 have been allowed.

(4) STATUS OF AMENDMENTS

The Office Action finally rejecting claims 1 and 11-23 was mailed on July 11, 2005. On November 8, 2005, Appellant telephoned the examiner to discuss the merits of the final rejection and the examiner explained the she will maintain the rejection based on Appellant's arguments about the phrase "... said differential assembly being rotatable relative to said ring gear ...". A Notice of Appeal was filed November 14, 2005. Subsequently, there have been

no other papers filed by the Appellant or issued by the U.S. PTO.

(5) SUMMARY OF CLAIMED SUBJECT MATTER

Figure 2 shows a ring gear sub-assembly 210 and a differential sub-assembly 220 of the torque transfer assembly of this invention, whereby the ring gear sub-assembly 210 is designed to rotate relative to the differential sub-assembly 220.

With reference to Figure 3, a speed-sensitive torque coupling assembly (shown generally as 240) is provided between the ring gear sub-assembly 210 and the differential sub-assembly 220 to regulate the difference in rotational speed between the sub-assemblies 210, 220. The speed-sensitive torque coupling assembly 240 included in the preferred embodiment of the present invention comprises a fluid pump 250 and a clutch pack 260. The fluid pump shown and described herein is a Gerotor type pump of the automatically reversible unidirectional flow type.

The friction clutch pack 260 includes sets of alternating outer friction plates 260a and inner friction plates 260b which are respectively fixed to the outer case member 214 and the clutch sleeve 270. The clutch sleeve 270 is in turn integrally formed with or splined to the differential case 222. The clutch plates 260a are adapted to frictionally engage the clutch plates 260b to form a torque coupling arrangement between the outer case member 214 and the differential case 222. Torque is transferred from the outer case 214 to the differential case 222, which drives the differential mechanism.

The clutch pack 240 is selectively actuated by a hydraulic clutch actuator including the speed sensitive hydraulic displacement pump 250, and a piston assembly 265 that axially loads the clutch pack 260.

The speed sensitive hydraulic displacement pump 250 disposed between the outer case member 214 and the differential case 222 actuates the clutch pack 260 when the relative rotation between these components 214, 222 occurs. It will be appreciated that a hydraulic pressure generated by the pump 250 is substantially proportional to a rotational speed difference between the outer case member 214 and the differential case 222.

Preferably, the hydraulic displacement pump 250 employed to provide pressurized hydraulic fluid to actuate the clutch pack 260 is a gerotor pump. The gerotor pump 250 includes an outer ring member, an outer rotor, and an inner rotor. The inner rotor is drivingly coupled to the clutch sleeve 270, and the outer ring member is secured to the outer case member 214. The inner rotor has one less tooth than the outer rotor and, when the inner rotor is driven, it will drive the outer rotor, which can freely rotate within the outer ring member eccentrically with respect to the inner rotor, thus providing a series of decreasing and increasing volume fluid pockets by means of which fluid pressure is created. Therefore, when relative motion takes place between outer case member 214 and the differential case 222, the inner rotor of the gerotor pump 250 generates hydraulic fluid pressure. However, it will be appreciated that any other appropriate type of hydraulic pump generating the hydraulic pressure in response to the relative rotation between the outer case member 214 and the differential case 222 is within the scope of the present invention.

The piston assembly includes a hydraulically actuated piston 265 which serves to compress the clutch pack 260 and retard any speed differential between the outer case

member 214 and the differential case 222. This results in a retardation of any speed differential between the front axle and rear axle. Pressurized hydraulic fluid to actuate the piston 265 and engage the clutch pack 260 is provided by the gerotor pump 250. In such an arrangement, when a speed difference exists, the hydraulic fluid is drawn into the pump 250 through a suction passage 290. The gerotor pump 250 pumps the pressurized fluid into a piston pressure chamber defined between the piston 265 and the piston housing to actuate the clutch pack 260. As the speed difference increases, the pressure increases. The pressurized fluid in the piston pressure chamber creates an axial force upon the piston 265 for loading the clutch pack 260, which is further resisted by the friction plates 260a and 260b. The loading of the clutch pack 260 allows for a torque transfer distribution between the outer case member 214 and the differential case 222.

A variable pressure relief valve assembly 300 is provided for selectively controlling a discharge pressure of the pump 250 and, subsequently, the clutch pack 260. When energized, solenoid-operated valve assembly 300 is capable of modulating a pump discharge pressure in a variable range from a minimum pressure to a maximum pressure, thereby variably controlling a drive torque distribution between the outer case member 214 and the differential case in a range from a minimum torque value to a maximum torque value.

The variable pressure relief valve assembly 300 according to the present invention, illustrated in detail in Fig. 4, is in the form of an electro-magnetic valve assembly and comprises a pressure relief check valve 332 controlled by an electro-magnetic actuator 334 that may be any appropriate electro-magnetic device well known in the art, such as solenoid.

The check valve 332 comprises a fluid relief passageway 336 that is in fluid communication with the piston pressure chamber, a substantially conical valve seat 338 that is

in open communication with the passageway 336, and a spherical valve closure member 340 adapted to seat in the valve seat 338 for sealing the fluid relief passageway 336. It will be appreciated that the valve closure member 340 may be in any appropriate form other than spherical, such as conical. The valve seat 338 is formed in the side wall 214a of the outer case member 214. The valve closure member 340 is movable between a closed position when the valve closure member 340 engages the valve seat 338 (as shown in Fig. 4), and an open position when the valve closure member 340 is axially spaced from the valve seat 338.

The electro-magnetic actuator 334 comprises a substantially annular coil housing 342, a coil winding 344 wound about the coil housing 342, and a substantially annular armature 352 axially movable in the direction of the rotational axis 11. The armature 352 is coaxial to the coil winding 344 and is radially spaced from the coil housing 342, thus defining an air gap 356. The annular armature 352 is supported within an armature bushing 354 for axially movement in the direction of the axis 11. The armature bushing 354 is non-rotatably mounted to the side member 214a of the outer case member 214 by any appropriate means, such as press-fitting, adhesive bonding, etc. Preferably, the armature bushing 354 is made of any appropriate non-magnetic material well known to those skilled in the art.

In the exemplary embodiment illustrated in Figs. 3 and 4, the armature 352 is disposed outside the coil winding 344 of the electro-magnetic actuator 334. Alternatively, the armature 352 may be disposed within the coil winding 344.

The valve closure member 340 is urged and held in place by against the valve seat 338 by an actuator plate 358. In turn, the actuator plate 358 is adapted to engage the armature 352 of the electro-magnetic actuator 334 disposed outside the coil winding 344 thereof.

When electrical current is supplied to the coil winding 344, a magnetic flux is caused to flow through the armature 352. The magnetic flux creates an axial force that axially displaces the armature 352 relative to the coil winding 344. The armature 352 moves the actuator plate 358, which, in turn, urges the valve member 340 upon the valve seat 338 with a predetermined axial retaining force that is a function of the electrical current supplied to the coil winding 344. It will be appreciated by those skilled in the art that the pressurized hydraulic fluid will not flow through the pressure relief valve 332 until the hydraulic pressure generated by the gerotor pump 326 results in a reaction force larger than the axial retaining force exerted to the armature 352 by the magnetic flux generated by the coil winding 344, thereby pushing the valve closure member 340 out of the valve seat 338.

When a maximum current is applied to the coil winding 344 of the solenoid actuator 334, the retaining force of the pressure relief valve 332 is at its maximum, thus a maximum release pressure is provided by the pressure relief check valve 332. In this configuration, the maximum pressure attainable within the outer case member 214 is sufficient to fully actuate the hydraulic clutch pack 260 which results in providing the limited slip function in the torque transfer assembly 10, and the limited slip feature is in the fully "ON" condition.

The pressure limit of the pressure relief valve 332, i.e. the release pressure of the pressure relief valve 332, can be adjusted by controlling the current applied to the coil winding 344 of the electro-magnetic actuator 334.

As less current is applied to the coil winding 344, less axial retaining force is exerted to the relief valve 332, thus the less is the release pressure provided by the relief valve 332. This results in an adjustment mechanism for lowering the maximum system pressure attainable within the outer case member 12.

When a minimum current is applied to the coil winding 344 of the solenoid actuator 334, the retaining force of the pressure relief valve 332 is at its minimum, thus a minimum release pressure is provided by the relief valve 332. In this configuration, the limited slip feature is in the fully "OFF" condition in that the maximum pressure which can be obtained in the outer case member 214 is not high enough to engage the clutch pack 260, essentially disabling the limited slip feature of the hydraulic torque transfer assembly without affecting the torque transfer capability.

In between the "ON" and "OFF" conditions of the torque transfer assembly the release pressure of the relief valve 332 may be set at any value by modulating the current applied to the coil winding 344 of the solenoid actuator 334. This provides the hydraulic torque transfer assembly 10 with a variable maximum pressure limit in which the amount of the limited slip available to the torque transfer assembly 10 can be limited and optimized to match various vehicle operating conditions. This provides an opportunity to dynamically control the hydraulic pressure for traction enhancement. For example, if the release pressure is set at a low value, a control system can be used to sense wheel speeds or speed differences and allow for increased hydraulic pressure. The increase in pressure available may be a function of the speed difference. This will result in an optimized amount of limited slip between the fully "ON" and "OFF" conditions.

(6) GROUNDS OF REJECTION TO BE REVIEWED ON APPEAL

Claims 1 and 11-23 stand rejected under 35 U.S.C. 102(e) as being anticipated by Kaplan et al. (USP 6,733,411) (hereinafter referred to as Kaplan et al. '411).

(7) ARGUMENTS

Sub-Paragraph (iii)

Claims 1 and 11-23 stand rejected under 35 U.S.C. 102(e) as being anticipated by Kaplan et al. '411. It is noted that claim 1 is an independent claim. It is also noted that claims 11-23 depend upon the base claim 1.

Regarding claim 1: The present invention is a torque transfer assembly where the differential assembly 220 rotates relative to the ring gear assembly 210 as shown in Figure 2. A torque coupling assembly 240 is disposed between the ring gear assembly 210 and the differential assembly 220. The claims recite a "variable pressure relief valve assembly" adapted to help control the torque coupling assembly 240. Claim 1 explicitly recites,

"...

a ring gear assembly adapted to deliver an input drive torque;

a differential assembly adapted to receive said input drive torque, said differential assembly being rotatable relative to said ring gear assembly;

..."

See claim 1 appended hereto.

Kaplan et al. '411 teaches a limited slip differential assembly where the ring gear and differential assembly are bolted together (see Figure 2 of Kaplan et al. '411). Thus, the ring gear cannot rotate relative to the differential assembly as set forth in the claimed invention. Applicant had previously amended claim 1 to clarify the structural distinction between the torque transfer assembly of this invention and the limited slip differential of Kaplan et al. '411.

In response to Appellant's amendment, the examiner stated that "the differential case of Kaplan et al. is rotatable and since the ring gear is bolted to the case, then the ring gear is rotatable relative to (with regard to) the differential case." See Final Rejection at page 3.

Therefore, the patent examiner asserts that two rotating members that are bolted together anticipate a limitation where the two members rotate independently of one another.

In a telephone discussion with the examiner, the examiner maintained the position set forth in the Final Rejection - a position that two rotating members that are bolted together rotate with regard to one another.

Appellant asserts that the examiner's position in this regard in untenable in light of the description in the present application and the common definition of two member that are relative rotatable. In other words, two members that are bolted together cannot also be rotatable relative to one another.

For these reasons, Kaplan et al. '411 does not meet the standard of anticipation and the rejection of claim 1 under 35 USC § 102(e) is improper.

Claims 11-23 introduce new limitations, and further defining the present invention over Kaplan et al. '411.

In view of the foregoing, it is respectfully submitted that this application is in condition for allowance, and notice to that effect is earnestly solicited. Appellant will request an oral hearing on the merits within two months after the date of the Examiner's answer.

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(8) APPENDIX OF CLAIMS ON APPEAL

1. A torque transfer assembly in a motor vehicle, said torque transfer assembly comprising:

a ring gear assembly adapted to deliver an input drive torque;

a differential assembly adapted to receive said input drive torque, said differential assembly being rotatable relative to said ring gear assembly;

at least one output shaft drivingly connected to said differential assembly through a differential gear mechanism;

a friction clutch pack for selectively delivering said input drive torque from said ring gear assembly to said differential assembly, said clutch pack comprising at least one inner friction plate coupled to rotate with said differential assembly and at least one outer friction plate coupled to rotate with said ring gear assembly; and

a hydraulic clutch actuator for selectively frictionally loading said clutch pack, said actuator comprising:

a hydraulic pump for generating a hydraulic pressure to frictionally load said clutch pack; and

a variable pressure relief valve assembly to selectively control said friction clutch pack, said variable pressure relief valve assembly including a valve closure member, a valve seat complementary to said valve closure member, and a electro-magnetic actuator for engaging said valve closure member and urging thereof against said valve seat so as to selectively vary a release pressure of said pressure relief valve assembly based on a magnitude of an electric current supplied to said electro-magnetic actuator,

wherein said valve closure member is movable between a closed position when said valve closure member engages said valve seat and an open position when said valve closure member is axially spaced from said valve seat.

2. The torque transfer assembly as defined in claim 1, wherein said electro-magnetic actuator including a coil winding supported by said ring gear assembly and an armature radially spaced from said coil winding and axially movable relative thereto in response to a magnetic flux generated by said coil winding when said electrical current is supplied thereto, said armature engages said valve closure member and urges thereof against said valve seat with an axial force determined by said magnitude of said electric current for selectively setting up said release pressure of said valve closure member.

3. The torque transfer assembly as defined in claim 2, wherein said coil winding is coaxial to an axis of rotation of said ring gear assembly.

4. The torque transfer assembly as defined in claim 2, wherein said coil winding is rotatably supported by said ring gear assembly.

5. The torque transfer assembly as defined in claim 2, wherein said coil winding is substantially annular in shape and is mounted substantially coaxially to an axis of rotation of said ring gear assembly.

6. The torque transfer assembly as defined in claim 2, wherein said coil winding is wound about a coil housing rotatably mounted to an outer peripheral surface of said ring gear assembly.

7. The torque transfer assembly as defined in claim 6, wherein said coil housing is substantially annular in shape and is mounted substantially coaxially to an axis of rotation of said ring gear assembly.

8. The torque transfer assembly as defined in claim 2, wherein said armature is non-rotatably coupled to said ring gear assembly.

9. The torque transfer assembly as defined in claim 2, wherein said armature is disposed outside said coil winding of said electro-magnetic actuator.

10. The torque transfer assembly as defined in claim 2, wherein said armature is substantially annular in shape and is mounted substantially coaxially to an axis of rotation of said ring gear assembly.

11. The torque transfer assembly as defined in claim 1, wherein said friction clutch assembly is a friction clutch pack including a plurality of inner friction plates coupled to rotate with said differential assembly and a plurality of outer friction plate coupled to rotate with said ring gear assembly, said friction plates being frictionally engageable with one another.

12. The torque transfer assembly as defined in claim 1, wherein said hydraulic pump is disposed within a housing defined by said ring gear assembly and generates a hydraulic pressure in response to relative rotation between said ring gear assembly and said differential assembly.

13. The torque transfer assembly as defined in claim 12, wherein said pump is a gerotor pump.

14. The torque transfer assembly as defined in claim 1, wherein said variable pressure relief valve assembly is adapted to selectively set a maximum hydraulic pressure attainable within said ring gear assembly between a maximum release pressure and a minimum release pressure.

15. The torque transfer assembly as defined in claim 1, wherein said hydraulic clutch actuator further including a piston assembly disposed within said ring gear assembly between said pump and said clutch pack and defining a pressure chamber, wherein said variable pressure relief valve assembly selectively controls a maximum hydraulic pressure attainable within said pressure chamber.

16. The torque transfer assembly as defined in claim 15, wherein said variable pressure relief valve assembly selectively controls said maximum pressure attainable within said pressure chamber between a maximum release pressure and a minimum release pressure.

17. The torque transfer assembly as defined in claim 16, wherein said minimum release pressure is at a level that prevents actuation of said friction clutch pack.

18. The torque transfer assembly as defined in claim 16, wherein said maximum release pressure is at a level that enables complete actuation of said friction clutch pack.

19. The torque transfer assembly as defined in claim 16, wherein said maximum hydraulic pressure attainable within said pressure chamber is adjustable between said minimum release pressure and said maximum release pressure so as to enable partial actuation of said friction clutch pack.

20. The torque transfer assembly as defined in claim 1, wherein said variable pressure relief valve assembly is adapted to selectively set a maximum hydraulic pressure attainable within said ring gear assembly between a maximum and a minimum release pressure.

21. The torque transfer assembly as defined in claim 20, wherein said minimum release pressure is at a level that prevents actuation of said friction clutch pack.

22. The torque transfer assembly as defined in claim 20, wherein said maximum release pressure is at a level that enables complete actuation of said friction clutch pack.

23. The torque transfer assembly as defined in claim 20, wherein said maximum hydraulic pressure attainable within said ring gear assembly is adjustable between said minimum release pressure and said maximum release pressure so as to enable partial actuation of said friction clutch pack.

24. The torque transfer assembly as defined in claim 2, wherein said armature has a substantially U-shaped cross-section.

25. The torque transfer assembly as defined in claim 2, wherein said coil winding is wound about a coil housing rotatably mounted to said ring gear assembly and wherein said armature is off-set from said coil housing to a distance that ensures that said axial force applied upon said valve closure member by said electro-magnetic actuator is substantially constant as said valve closure member moves from said closed position to said open position and said axial force is a function only of said electrical current supplied to said coil winding.

(9) EVIDENCE APPENDIX

Not applicable

(10) RELATED PROCEEDINGS APPENDIX

Not applicable